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CO₂ Stirling Heat Pump for Residential Use

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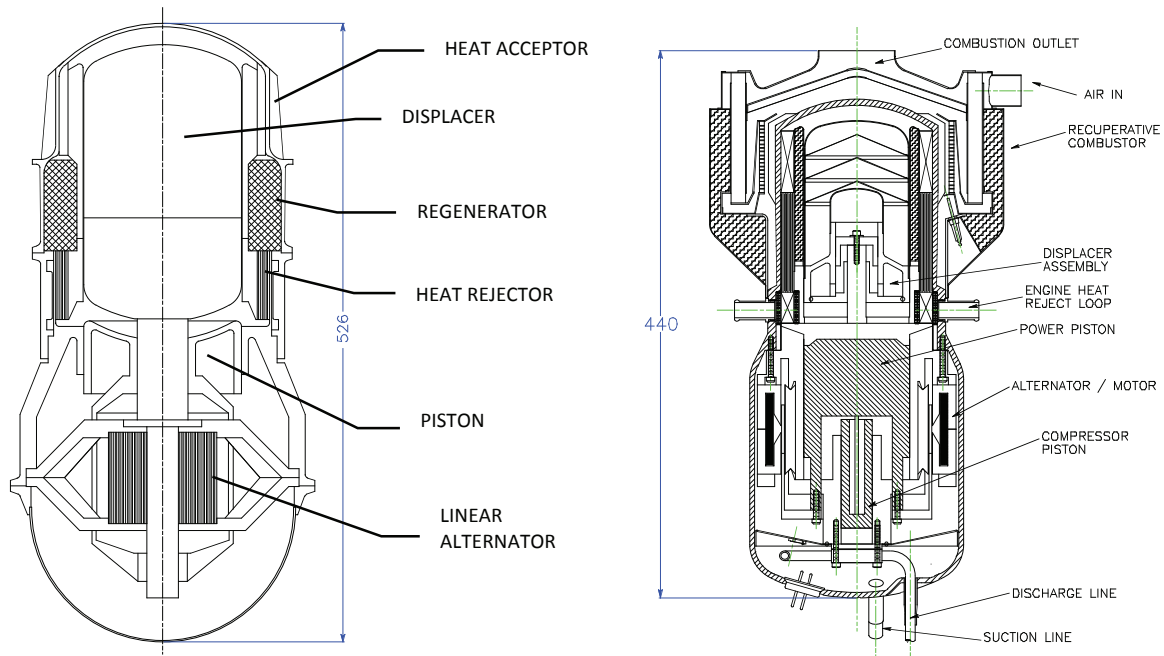
ABSTRACT

The free-piston Stirling engine (FPSE) is a reliable, high-efficiency small-capacity power generator often used in conjunction with a linear alternator. In this work, the FPSE concept is extended to directly drive a compressor by way of a common piston structure. The working gas is CO₂, which freely mixes between the engine and the transcritical compressor cycle. Typically, FPSE have used helium as the working medium. The use of CO₂ compromises the engine efficiency by about 10%. However, the resulting mechanical simplicity by not having to provide a hard separation between the working medium of the engine and compressor is compelling. Calculations are presented for US, Dutch and Japanese conditions using natural gas. It is shown that extremely high primary energy ratios are possible concomitant with reductions in CO₂ emissions, particularly when coupled to a ground source. Since the device is externally heated, biomass is a possible fuel source and an example calculation is included.

1. INTRODUCTION

The question is often raised as to what is the most effective use of primary energy for heating or cooling. It is well known that a heat pump will deliver more energy than the primary input because it is able to extract energy from the environment and deliver that to the sink together with the primary input. However, most heat pumps use electrical energy where the heat of power generation at the central power station is not usually recovered. If the heat of power generation could be recovered and added to the heat energy delivered by the heat pump, then overall energy utilization will be improved. One method to achieve this is by using an engine driven heat pump where the rejected heat from the engine is recovered for heating. Unfortunately most engines are short-lived and too inefficient to provide much benefit. An exception is the FPSE, which has demonstrated both long-life and high efficiency (Wood & Carroll, 2004). Typically, FPSEs deliver electrical power by way of an internal linear alternator as shown in Figure 1a. In concept, the FPSE would be an ideal means to drive a heat pump. Gas fired heat pumps using this principle have been pursued before. Efforts include a Duplex Stirling arrangement by Sunpower in 1983 (Penswick & Urieli, 1984), a free-piston Stirling engine hydraulically driving a Rankine heat pump by Mechanical Technology Inc. (Marusak & Ackermann, 1985), a second and third effort by Sunpower using an inertial drive and magnetic coupling to a Rankine heat pump (Wood, Unger, & Lane, 2000), (Chen & McEntee, 1993) and various efforts in Japan (MITI, NEDO, 1986) and Europe (Lundqvist, 1993). All of these efforts identified the obvious advantages of the Stirling engine. Namely, its high part load efficiency coupled with a potential for high reliability and long life. However, these efforts failed to produce a practical, cost-effective device mainly due (in the case of the Rankine – Stirling machines) to the perceived need to isolate the working medium of the engine and the heat pump. The concept presented here offers a simple technique that allows direct coupling of a FPSE to a CO₂ heat pump while preserving the virtues of the FPSE (Berchowitz & Kwon, 2005). It is referred to as a Free-Piston Stirling Heat Pump (FPSHP). The central idea is to replace the working gas of the FPSE (typically helium) with CO₂ and to couple the

engine piston directly to the piston of a CO₂ heat pump as shown in Figure 1b. By so doing, the problem of isolating the engine working gas from the heat pump is completely avoided. While CO₂ is not an ideal fluid for Stirling engines and there is an efficiency penalty of about 10%, the enormous simplification possible to an otherwise complicated device has overwhelming merit.



Figures 1: a) Free-piston Stirling engine with linear alternator capable of about 1 kW(e). b) Free-piston Stirling heat pump capable of about 10 kW of heat delivery (Dimensions in mm.).

2. BASIC THEORETICAL CONSIDERATIONS

2.1 Simple Theory

A simple analysis based on fundamental principles is developed in order to show some basic characteristics of the fuel or heat driven heat pump represented by Figure 2. In the theory developed here, all rejected heat is assumed to be recoverable, in principle. This is not strictly true; most practical systems will suffer some irrecoverable losses.

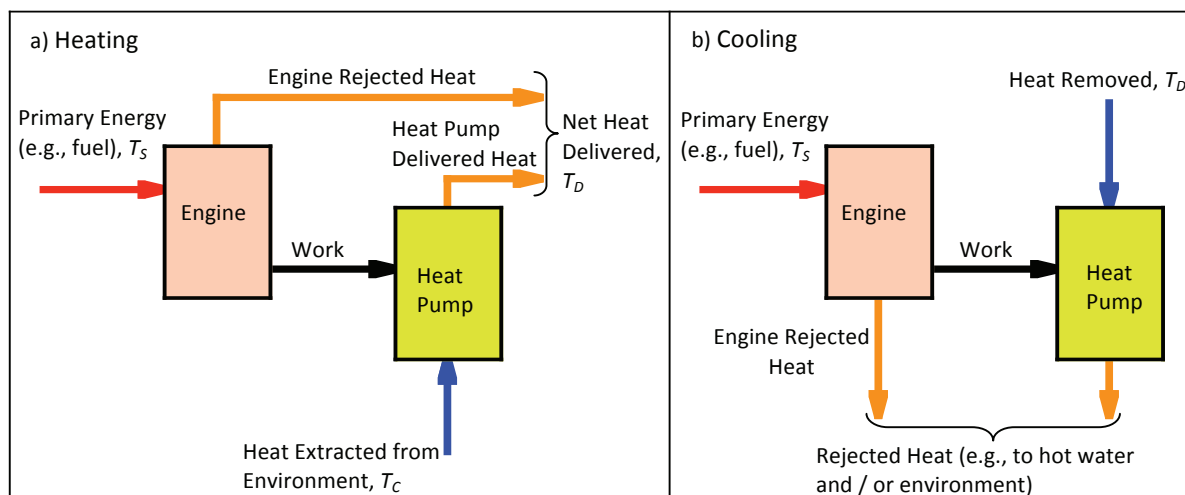


Figure 2: Schematic of an engine driven heat pump in a) heating and b) cooling modes.

The performance of a heat pump is defined by the Coefficient of Performance as follows:

$$\text{COP} = \frac{\text{Heat Energy Delivered or Removed}}{\text{Input Energy}} \quad (1)$$

For simplicity, the COP will be assumed to be proportional to the well-known Carnot COP, which is the ideal maximum. Thus:

$$\text{COP} = k_{\text{COP}} \cdot \text{COP}_{\text{Carnot}} \quad (2)$$

Where, for heating:

$$\text{COP}_{\text{Carnot}_H} = \frac{T_H}{T_H - T_L} = \frac{\tau_{\text{HP}}}{\tau_{\text{HP}} - 1} \quad (3)$$

And for cooling:

$$\text{COP}_{\text{Carnot}_C} = \frac{T_L}{T_H - T_L} = \text{COP}_{\text{Carnot}_H} - 1 = \frac{1}{\tau_{\text{HP}} - 1} \quad (4)$$

T_H and T_L being the high and low absolute temperatures of the cycle, respectively and τ_{HP} is the temperature ratio T_H/T_L . For the example shown in Figure 2, $T_H = T_D$ and $T_L = T_C$.

If the input for the heat pump is provided by an engine where the engine reject heat is offered as additional heat to the sink as shown in Figure 2a, then the component performances of such a system may be defined by:

The engine efficiency:

$$\frac{\text{Work}}{\text{Input Energy}} = k_{\eta} \eta_{\text{Carnot}} = k_{\eta} \frac{T_H - T_L}{T_H} = k_{\eta} \frac{\tau_e - 1}{\tau_e} \quad (5)$$

Where η_{Carnot} is the Carnot efficiency, k_{η} is the constant of proportionality and τ_e is the engine temperature ratio. Again, for the example of Figure 2, $T_H = T_S$ and $T_L = T_D$.

For the engine, the reject heat ratio is given by:

$$\frac{\text{Reject Energy}}{\text{Input Energy}} = 1 - k_{\eta} \eta_{\text{Carnot}} \quad (6)$$

Using Equations (2), (5) and (6) and noting that the engine work is the input energy for the heat pump, a heat-driven performance is given by:

$$\frac{\text{Net Heat Delivered}}{\text{Input Energy}} = \text{PER}_{\text{Heating}} = 1 + k_{\eta} \eta_{\text{Carnot}} (k_{\text{COP}} \cdot \text{COP}_{\text{Carnot}_H} - 1) \quad (7)$$

Where $\text{PER}_{\text{Heating}}$ is the primary energy ratio for heating.

In terms of temperature ratios, Equation (7) gives the following result:

$$\text{PER}_{\text{Heating}} = 1 + k_{\eta} \frac{\tau_e - 1}{\tau_e} \left(k_{\text{COP}} \frac{\tau_{\text{HP}}}{\tau_{\text{HP}} - 1} - 1 \right) \quad (8)$$

Where τ_{HP} is absolute temperature ratios of the heat pump.

For the ideal case, $k_\eta = k_{\text{COP}} = 1$, giving:

$$\text{PER}_{\text{Ideal Heating}} = 1 + \left(\frac{1}{\tau_e} \right) \frac{\tau_e - 1}{\tau_{\text{HP}} - 1} \quad (9)$$

Assuming $T_C = 5^\circ\text{C}$, $T_D = 40^\circ\text{C}$ and $T_H = 600^\circ\text{C}$, the ideal heating PER is 5.9 which implies that an ideal heating system lifting heat from a 5°C source and accepting primary energy from a 600°C source will deliver 5.9 units per unit input of heat energy to a sink at 40°C .

The results for the cooling example follow similarly. Except, in this case, the rejected heat may be used in one of three ways as shown in Figure 2b, namely, recovery, rejected to the environment or a combination of both. For the case where all the heat is recovered (engine + heat pump), the following result is obtained:

$$\text{PER}_{\text{Cooling}_{e+\text{HP}}} = 1 + k_\eta \eta_{\text{Carnot}} \left[k_{\text{COP}} (2\text{COP}_{\text{Carnot}_C} + 1) - 1 \right] = 1 + k_\eta \frac{\tau_e - 1}{\tau_e} \left[k_{\text{COP}} \left(\frac{\tau_{\text{HP}} + 1}{\tau_{\text{HP}} - 1} \right) - 1 \right] \quad (10)$$

When the heat pump reject heat is not recovered, as may be more typical since there is often a need to balance the annualized energy extracted from the ground, the result becomes:

$$\text{PER}_{\text{Cooling}_e} = 1 + k_\eta \eta_{\text{Carnot}} (k_{\text{COP}} \cdot \text{COP}_{\text{Carnot}_C} - 1) = 1 + k_\eta \frac{\tau_e - 1}{\tau_e} \left(k_{\text{COP}} \frac{1}{\tau_{\text{HP}} - 1} - 1 \right) \quad (11)$$

When neither engine nor heat pump reject heat is recovered, the result is:

$$\text{PER}_{\text{Cooling}} = k_\eta k_{\text{COP}} \cdot \eta_{\text{Carnot}} \text{COP}_{\text{Carnot}_C} = k_\eta k_{\text{COP}} \frac{1}{\tau_e} \frac{\tau_e - 1}{\tau_{\text{HP}} - 1} \quad (12)$$

For the result expressed by Equation (11), it is worth noting that the energy returned to the ground during cooling is available for extraction during the heating season. A lower τ_{HP} for heating would account for this effect.

2.2 Theoretical Results

The temperature ratio of the engine should be as high as possible for maximum efficiency. However, this is limited by material considerations. The hot end of the engine may be in the region of 550°C to 600°C and the reject side at around 50°C to 70°C (Hargreaves, 1991). A reasonable number for τ_e is therefore about 2.5. A well-designed recuperative burner should be able to achieve efficiencies of 80 to 90%. A similarly well-designed Stirling engine should manage a fraction of Carnot of between 0.55 to 0.70 giving an overall k_η of between 0.44 and 0.63 (Wood & Carroll, 2004 and Hargreaves, 1991). The overall k_{COP} for heat pumps vary from about 0.25 to 0.45 (IEA Heat Pump Centre, 2008). Using this data, the heating PER has been plotted against the temperature ratio for the heat pump as shown in Figure 3. Increasing temperature ratio represents a heat pump operating over a wider temperature span. Ground source heat pumps operate at reduced τ_{HP} , usually less than 1.1, while air source heat pumps operate at higher τ_{HP} , usually above 1.1. Given the assumed performances, an engine-driven ground source heat pump clearly has the capability of delivering a heating PER of better than 1.5 and may even approach values of 2.0 or more. Also shown, is the heating PER versus the heat pump temperature ratio for improved engine performance. There may be a strong cost multiplier associated with improved engine performance since this implies higher precision and more extensive heat exchangers. From Figure 3 it is clear that improved engine performance is not as valuable at higher τ_{HP} as it is at lower τ_{HP} . An increased engine performance from $k_\eta = 0.5$ to 0.6 (a 20% increase) gives about the same return at $\tau_{\text{HP}} = 1.1$ that a heat pump improvement of $k_{\text{COP}} = 0.35$ to 0.40 would do (a 14% increase). On the other hand, reducing τ_{HP} as may be achieved by improved thermal coupling with the source and sink, has a far stronger effect on overall performance. Clearly, as τ_{HP} increases, the engine driven heat pump approaches the performance of a 100% efficient furnace since in the limit, assuming all heat is captured, the heating COP cannot be less than 1.0.

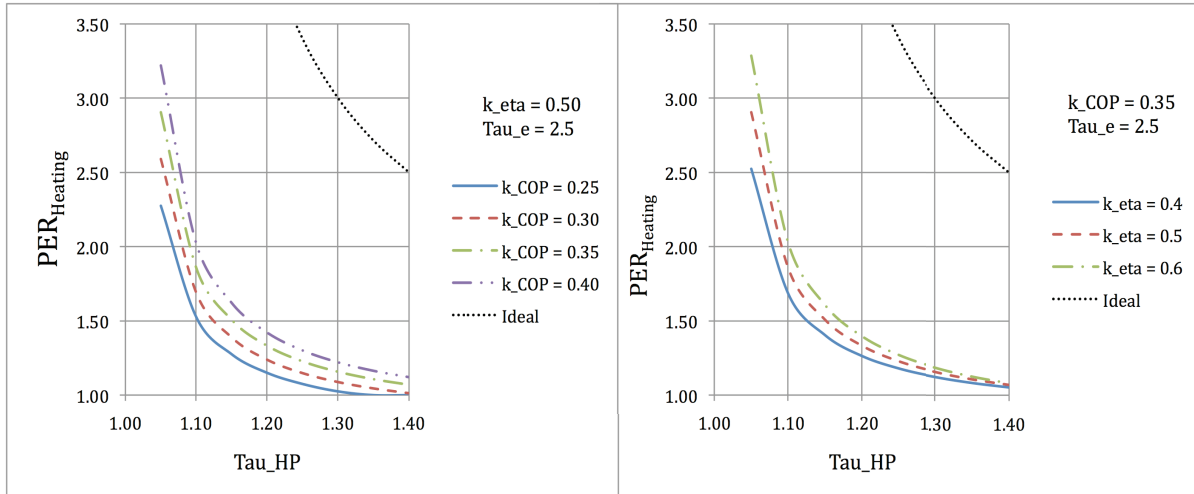


Figure 3: Heating PER versus heat pump temperature ratio of different values of k_{COP} and k_{η}

For the cooling case only Equation (11) is considered since, as mentioned, it is more typical not to recover the heat pump rejected heat. Again, air source systems tend to operate at heat pump temperature ratios above 1.1 while ground source systems operate below this number. Air source CO_2 systems become transcritical in cooling mode and tend to lose efficiency. For these systems, a constant k_{COP} with temperature ratio is not a good assumption. However, as ground source systems, the CO_2 heat pump can be designed to operate in Rankine mode during cooling. From Figure 4, it is clear that the low heat pump temperature ratio possible with ground source systems provides excellent overall PER even when the heat pump performance (k_{COP}) is low. Indeed, it is possible for the heat pump temperature ratio to approach 1 and even reach a condition where it is preferable to simply circulate a secondary heat transfer medium between the ground source (actually a sink) and the space being cooled. Note that as τ_{HP} increases, the PER eventually falls below 1 because in this cooling mode, not all of the rejected heat is recovered.

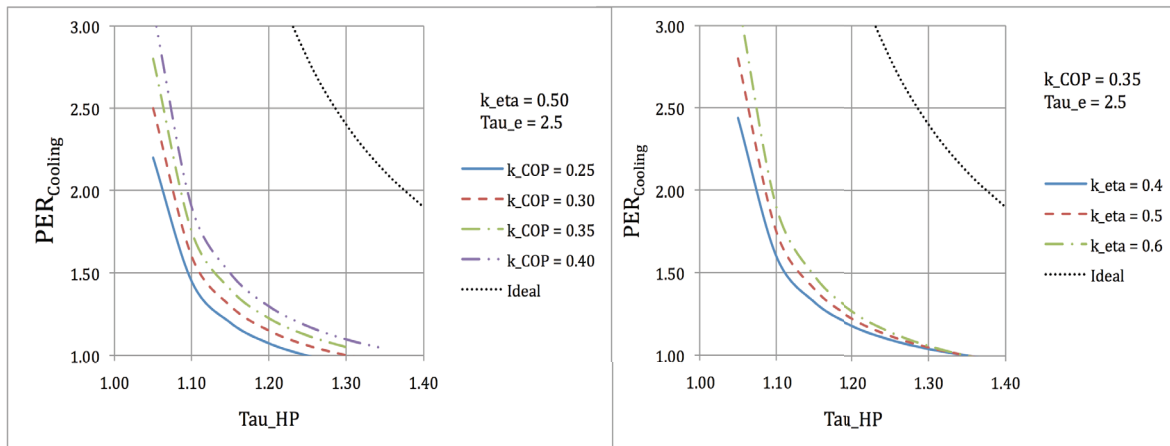


Figure 4: Cooling PER versus heat pump temperature ratio of different values of k_{COP} and k_{η} , Equation (11)

3. SOME SYSTEM EXAMPLES

A detailed system calculation has been made by Berchowit and Kwon (2005) for a US gas-fired system and is presented in Table 1. This system was designed for a ground water loop according to specifications set out in ARI Standard 325/330-1998. The burner blower and pumps were assumed to consume 225 W of grid power with an assumed PER of 0.38. No allowance was made for indoor air handling. In this example, $\tau_e = 2.7$, $\tau_{HP} = 1.08$ (heating), $\tau_{HP} = 1.02$ (cooling), $k_{\eta} = 0.41$, $k_{COP} \approx 0.49$ (heating) and $k_{COP} \approx 0.24$ (cooling). These numbers are based on average indoor air and ground water temperatures. Deployment of the ground source FPSHP would reduce CO_2 emissions by about 40 to 50% wherever a gas furnace was replaced. Where electric heat pumps are replaced by gas-

fired FPSHPs, the CO₂ offset is determined comparing to what would have been generated by the central power plant (the US national average is about 0.57 kg/kWh). CO₂ generation when combusting natural gas is about 56 to 57g/MJ. This translates to a savings of about 30 to 40% against a ground source electric heat pump operating with identical COP as in Table 1, a motor efficiency of 85% and gas-fired hot water.

Table 1: System performance calculated for a US homes – ground water source (Berchowitz & Kwon, 2005)

Component	Description	Heating	Cooling
FPSE	Fuel input, lower heating value of gas [W]	4,840	2,380
	Burner efficiency [%]	87	87
	Engine efficiency [%] at $T_H = 900$ K, $T_L = 330$ K	30	30
	Exhaust heat [W] partly using for super heating	630	310
	Engine reject heat [W] fully recovered	2,950	1,450
	Engine mechanical power [W]	1,260	620
CO ₂ cycle	Compressor overall efficiency (%)	90	91
	Cycle COP: ideal COP x compressor efficiency	6.57	12.18
	Condensation [W] (Rankine mode)	8,300 ($T_H = 303$ K)	9,000 ($T_H = 295$ K)
	Evaporation [W]	7,700 ($T_L = 280$ K)	7,500 ($T_L = 290$ K)
	Delivered/removed energy to/from indoor [W]	8,300	7,500
System performance	Delivered energy to water heating [W]	2,950	1,450
	PER with 2 pumps & burner blower	2.15	3.34

Table 2 summarizes calculations for Dutch homes for heating modes only. Owing to higher delivery temperatures, the heat pump cycle is transcritical. Here $\tau_e \approx 2.8$, $\tau_{HP} = 1.08$ to 1.12 (based on average source and sink temperatures), $k_\eta = 0.47$ and $k_{COP} \approx 0.35$. The lower k_{COP} is partly due to the sink being a secondary heat transfer fluid between the heat pump and the indoor air as is the case in hydronic systems.

Table 2: Heating only system performance calculated for Dutch new homes (Berchowitz, 2005)

Performance Parameters		Operation Modes: Water Source (10 °C → 7 °C) and Brine Source (0°C → -3 °C)					
		Space Heating + Hot Water		Hot Water Only		Space Heating Only	
		Water	Brine	Water	Brine	Water	Brine
FPSE $T_H = 900$ K $T_L = 320$ K	Burner Eff. [%]	87.0	87.0	87.0	87.0	87.0	87.0
	Fuel Input [W]	4977.0	5854.0	5322.7	6106.7	5525.1	6385.4
	Blower Power [W]	25.0	25.0	25.0	25.0	25.0	25.0
	Engine Eff. [%]	35.0	35.0	35.0	35.0	35.0	35.0
	Engine Power [W]	1515.5	1782.5	1620.8	1859.5	1682.4	1944.4
	Engine Reject Heat [W]	2814.5	3310.4	3010.0	3453.3	3124.4	3610.9
CO ₂ Cycle $T_L \approx 272$ to 282 K $T_H \approx 305$ K	Heating COP	4.83	3.80	4.38	3.54	4.12	3.30
	Heating Capacity [W]	7319.9	6779.0	7094.8	6585.2	6924.7	6435.2
	Indoor HX Pump [W]	25.0	25.0	25.0	25.0	25.0	25.0
	Source HX Pump [W]	50.0	50.0	50.0	50.0	50.0	50.0
Total Heat Capacity [W]		10134.4	10089.4	10104.8	10038.5	10049.1	10046.1
System Heating PER		1.96	1.67	1.83	1.59	1.76	1.53

A second Dutch application study, using biomass as the fuel source, is shown in Figure 5. The space heating performance curve for this example is shown in Figure 6 (van Rooijen *et al.* 2008). Here the burner efficiency is somewhat lower than that in Tables 1 and 2 partly due to the additional energy needed to gasify the biomass pellets. The system parameters come out to $\tau_e = 2.7$, $\tau_{HP} = 1.085$ (based on average source and sink temperatures), $k_\eta = 0.38$ and $k_{COP} \approx 0.31$. About 50% of the burner lost heat is recovered by the use of an exhaust heat recovery heat exchanger. The van Rooijen *et al.* study found a net CO₂ emissions savings against electric heat pumps of about 13% when using gas and about 38% when using biomass despite the much lower CO₂ emissions of Dutch central power generation (0.325 kg/kWh).

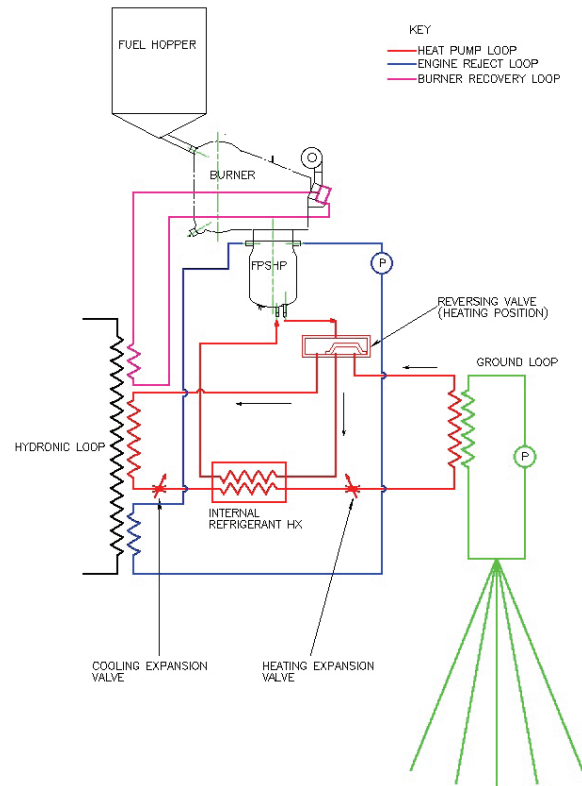
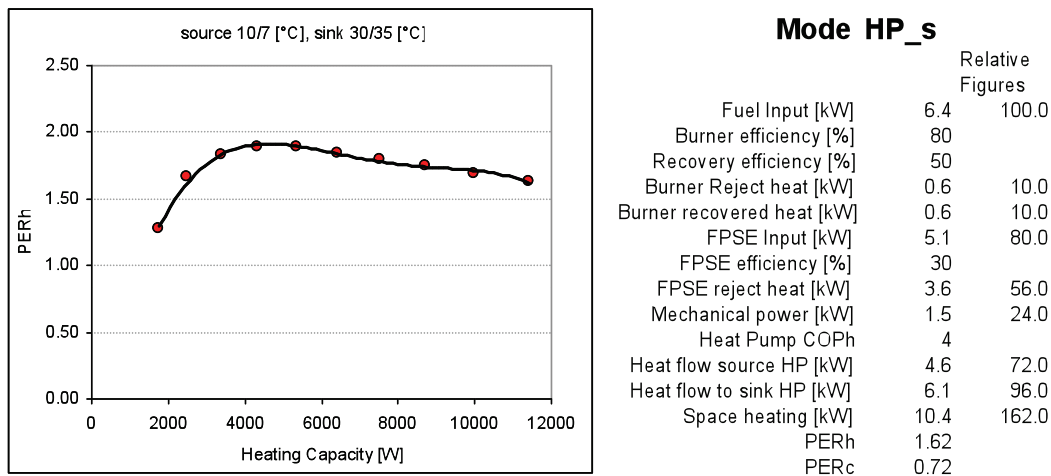
Figure 5: Layout of biomass system (van Rooijen *et al.* 2008)

Figure 6: Heating PER versus total heat delivered for the biomass system operating in space heating mode

A final example is a comparison done by Tezuka *et al.* (2007) against a current Japanese hot water heating system based on an electrically driven CO₂ transcritical cycle (though a number of companies manufacture the device, it is generally referred to as 'Ecocute'). Figure 7 shows the basic elements and energy flows of the system. The FPSHP was shown to have a substantially superior PER to the electrical heat pump (2.05 compared to 1.45) even though Japanese electricity is produced with high efficiency. In addition, nuclear plants generate a large fraction of Japanese electricity and therefore overall CO₂ emissions per kWh are low. Despite this, the gas-fired FPSHP is expected to have similar or slightly reduced CO₂ emissions compared to the electrically driven Ecocute.

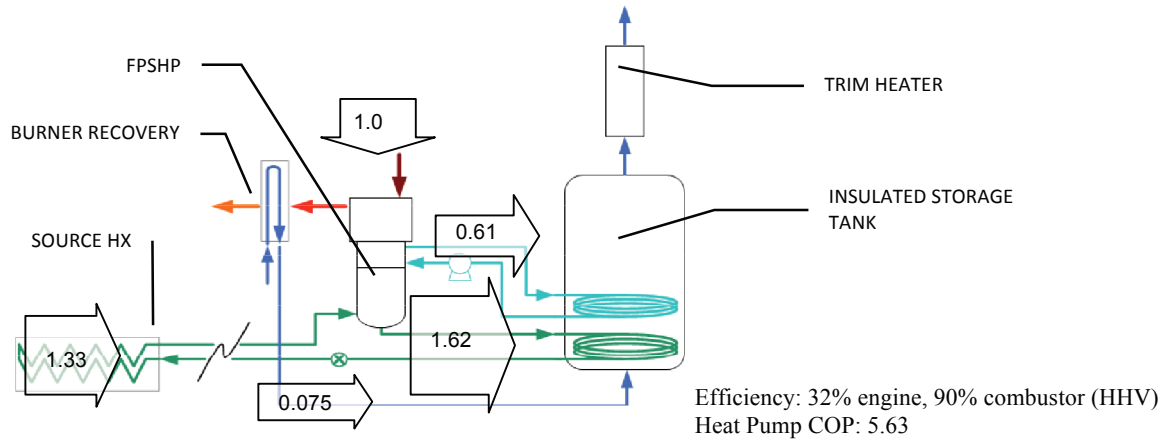


Figure 7: System elements and estimated energy flows for Japanese hot water system

4. CONCLUSIONS

Engine-driven heat pumps offer the possibility of high primary energy efficiency in HVAC applications where engine waste heat may be utilized effectively. Using simple thermodynamic considerations, it is shown that the PER sensitivity to temperature and performance can be characterized by a few parameters that are fairly easily established. Among these parameters it is clear that the heat pump temperature ratio has the strongest influence on overall primary energy ratio; confirming the superiority of ground source over air source heat pumps in terms of efficiency. In addition, there is obviously an ideal upper performance limit associated with engine or heat driven heat pumps that represents a fundamental maximum that may be used as a metric in comparing real systems. The proposed free-piston Stirling heat pump offers one practical possibility of realizing such a system.

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